

A high pressure and high frequency diaphragm engine: Comparison of measured results with thermoacoustic predictions



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HIGHLIGHTS

- High frequency and high pressure diaphragm engine delivers good power density.
- Engine has no sliding seals and thus no wear or seal leakage leading to long life.
- No high tolerance or exotic parts leading to low cost.
- Good agreement of thermoacoustic model with experimental results.
- Prototype at 500 °C has 21% efficiency and 580 W output.

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ABSTRACT

A novel diaphragm Stirling/thermoacoustic engine has been developed and tested that operates at high pressure and high frequency thereby delivering good power density and efficiency. This engine does not require any high tolerance or exotic parts and may thus be amenable to low cost construction in volume. Given the high frequency (500 Hz) and high working gas pressure (90 bar He) the inertia of the working gas is not negligible and thus traditional Stirling engine analysis fails to properly model such an engine. Instead, the engine is successfully modeled as a traveling wave thermoacoustic engine with a mechanical resonator (the displacer) closing the acoustic power loop. The predictions of the thermoacoustic model are compared with experimental results obtained from a prototype engine.

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1. Introduction

Stirling engines have not enjoyed much commercial success to date. One factor in this lack of success is the difficulty of making reliable, low cost, sliding gas seals without the use of lubricants. This problem may be avoided by building an engine based on flexing diaphragms rather than pistons. At least one such engine was built in the 1970s at Atomic Energy Research Lab [1–5]. This engine, called the thermo-mechanical generator (TMG), was however only capable of very modest (~ 70 W) power output and had very low power density. The low power density was due to a combination of unpressurized working gas and low frequency operation. It did however demonstrate the simplicity and reliability of the metal flexure based approach, running for 100,000 h maintenance free. A later modestly pressurized version produced about 170 W of indicated power [2]. The only other pure metal flexure based Stirling engine found in a literature search was patented by Mechanical Technologies Inc. [6] but no performance details are publicly

available. Mechanical Technologies Inc. additionally developed a hybrid Stirling engine with a diaphragm power transducer but with a non-diaphragm, sliding and electrically driven displacer. Published performance data indicate that it produced on the order of 2 kW of mechanical power [7,8]. However, it was not a pure diaphragm engine given the driven, sliding displacer and as such did not realize all the advantages of a pure flexure based engine. In order to make the power density of a pure flexure engine competitive with a piston Stirling engine it is necessary to increase both the operating pressure and frequency by an order of magnitude or more in order to compensate for the inherent small stroke of the flexures. At high pressures and frequencies the inertia of the working gas is no longer negligible, thus a thermoacoustic model is convenient for complete understanding of the engine operation. Thermoacoustic theory has been successfully employed for heat engine analysis by a number of authors [9–13].

A high power density flexure based engine has been designed and built along with a matching thermoacoustic computer model. The engine design is a marriage of a piston-less thermoacoustic engine and a beta free-piston Stirling engine. Flexing diaphragms, akin to the loud speakers used as acoustic drivers in some thermoacoustic machines, replace the pistons of a traditional Stirling

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engine [2] and a mechanical resonator, the displacer, replaces the acoustic feedback network of the traditional traveling wave thermoacoustic engine [10]. The primary distinguishing feature of this engine is the lack of sliding seals. In addition to the obvious disadvantages of leakage, possible wear and losses there are cost disadvantages to sliding seals. Sliding seals without lubricants often require close tolerance fits with the dimension of mating parts matched to on the order of $10\text{ }\mu\text{m}$. This requires costly precision machining and quality assurance and often manual sorting or polishing of parts to ensure good matches. In comparison, in this flexure engine dimensional machining tolerances are about of an order of magnitude larger and more crucially it is not necessary to closely match one machined part to another so parts may be used as machined. For example, the power transducer flexure is a steel diaphragm with a specified profile in order to keep bending and pressure stresses as uniform as possible. At the thinnest point near the periphery the transducer has a thickness of about 1 mm . It thickens gradually with decreasing radius reaching about 15 mm at the center. The required thickness tolerance as a function of radius is no more than on the order of $100\text{ }\mu\text{m}$. Any small deviation in profile leads only to slight changes in stiffness and hence operating frequency but does not have any effect on sealing or wear.

As an additional distinguishing feature in this engine there is no need to contain the linear alternator within the pressure vessel. Due to the small stroke of the diaphragm it is possible to seal the engine with a separate flexure (spring tube 104 in Fig. 1). In addition to hermetically sealing the engine, while allowing transmission of the vibration to outside the pressure vessel, it acts as the primary mechanical spring in the system. Hermetic sealing allows the linear alternator to be outside the pressure vessel without causing working gas leakage and this leads to further pressure vessel size and weight reduction. Since the alternator is outside, and no acoustic resonator is needed, the pressure vessel is very compact.

A flexure based engine of the type described in this paper is likely scalable for powers in the range 100 W to about 10 kW but no detailed modeling or design has yet been done for any engine with greater than 3 kW or less than 500 W of output. Preliminary bottom up costing of the engine taking into account the material and likely manufacturing method of each part in the engine indicates that a 1 kW engine weighing 30 kg can be built for less than

$1\$/\text{W}$ in moderate volume. The result is a compact engine without the problems associated with sliding seals. The small size of the pressure vessel leads to excellent engine power density. The inherent simplicity of the design, easy manufacturability and small size may lead to extremely high reliability and low manufacturing costs per watt of output.

2. Design of the engine

A schematic diagram of the diaphragm engine is shown in Fig. 1 and a sectioned rendering of the engine as built is shown in Fig. 2. The engine is filled with 90 bar helium and operates in a beta configuration. The drive shaft (113), depicted at the bottom, is coupled to the (power piston) diaphragm (103) through a folded spring tube (104). The dual flexure structure (111, 112) above the diaphragm acts as the displacer. The action of the displacer is the result of its tuned mechanical resonance interacting with the gas dynamics. Acoustic power is created in the compression space (102) between the diaphragm and the displacer by their out of phase motion. The motion of the displacer leads the motion of the diaphragm by approximately 50 degrees. The acoustic power travels radially along the diaphragm surface and is piped through 48 small access tubes (106) to 24 cylindrical heat exchanger and regenerator (109) units distributed around the outer periphery of the engine. Next the acoustic power flows through the water cooled cold heat exchanger (108) which is made from carbon velvet. It then continues through the glass micro-capillary array regenerator before passing through the hot heat exchanger (110) which is also constructed from carbon velvet. The acoustic power is amplified passing through the regenerator, due to the temperature gradient across the regenerator (maintained by the adjacent heat exchangers), and travels into the expansion volume (101) where it is absorbed by displacer motion and coupled back to the compression volume through the action of the displacer. Due to amplification by the regenerator the returned acoustic power is greater than the initial power and the excess power is transferred to a linear alternator (not shown) by the drive shaft (113).

Below the diaphragm, as shown in Fig. 1, is an enclosed volume called the bounce space (105) which is filled to the same pressure as the engine working volume. The role of the bounce space is to balance the static pressure of the working gas across the diaphragm. This volume is sealed from the atmosphere by a pressure vessel (100) which includes a spring tube that is coupled to the diaphragm. The spring tube is a very stiff spring and is the dominant spring contribution to the mechanical resonance of the diaphragm, drive shaft, and alternator system. The spring tube is of a folded

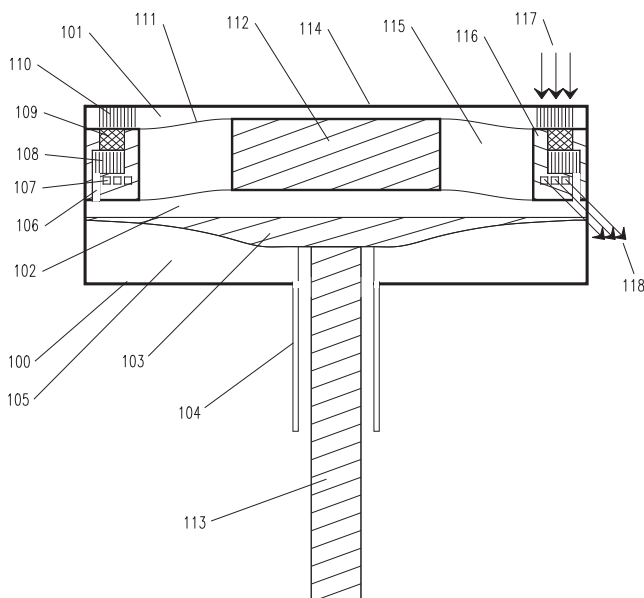


Fig. 1. Diaphragm engine schematic.

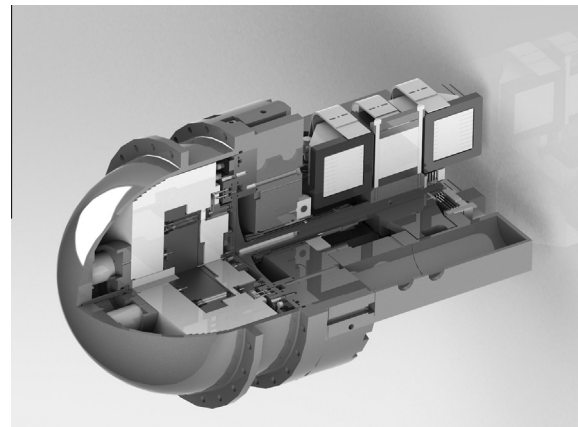


Fig. 2. Cross-sectioned rendering of the diaphragm engine.

design that results in a long effective length in a compact form. The length ensures that the material stresses are lower than the infinite fatigue limit. The total diaphragm spring system is a combination of the spring tube, the diaphragm, the drive shaft (113), magnetic interactions in the alternator (not shown), and the gas spring of the engine volume. The total spring constant in combination with the total mass of this system combine to set the operating frequency. The bounce volume is made much larger than the engine working volume and therefore has much smaller pressure amplitude than the gas in the engine. Given the very stiff tube spring there is no need for an additional gas spring with additional inherent losses on the bounce space side of the diaphragm.

The 140 mm diameter stainless steel diaphragm has a tapered profile, along the bounce volume side, such that it is thicker in the middle and thins toward the edge. The shape is designed to maintain uniform stress throughout the flexing region of the diaphragm taking into account both the deflection stresses and the pressure bulge stresses. Diaphragm stresses are kept below infinite fatigue limits. With the bounce volume pressurized to the mean pressure, the diaphragm only needs to handle the cyclic pressure amplitude in the engine working volume. The nominal gap between the diaphragm and the displacer is on the order of 300 μm while the displacer and diaphragm center amplitudes are on the order of 200 μm . The working gas side of the diaphragm is also profiled such that at maximum excursion toward the displacer, the surface is flat. The chamber heights and diaphragm deflections are highly exaggerated in Fig. 1 for visibility.

The carbon velvet cold heat exchangers (labeled 108 in Fig. 1) are approximately 10 mm in diameter and on the order of 500 μm thick with the fibers aligned collinearly with the cylindrical axis. The individual fibers have a nominal diameter of 10 μm , conductivity of 800 W/mK and are randomly distributed with a packing fraction of 0.05–0.10 by volume. The fibers form a random pin fin array heat exchanger. One face of the material is thermally anchored to a water cooled ($\approx 15^\circ\text{C}$) copper heat sink (107). The other side faces the regenerator. The gas flow in the heat exchangers is predominantly in the radial direction with some axial flow near the regenerator. The carbon velvet has very high thermal conductivity and the spacing of the fibers provides good thermal contact with the gas. The hot heat exchangers (110) are arranged similarly.

The 24 regenerators are made using soda lime glass micro-capillary arrays with a diameter of approximately 10 mm matching the size of the heat exchangers. The gas passage pores have nominal diameters of 29 μm and are arranged in a hexagonal array such that 45% of the cross-sectional area is open. The maximum operating temperature of the engine is limited to 500 $^\circ\text{C}$ by the softening point of the soda-lime glass. The regenerators are enclosed by 0.15 mm wall thickness Inconel 625 tubes. These tubes provide structural strength as well as sealing the working gas from the surrounding pressure vessel gas.

In this prototype design, the engine is heated using 24 cartridge heaters with 250 W power rating each. The hot side of the engine is capped by a ~ 40 mm thick Inconel 625 plate (the “hot cap”, labeled 114 in Fig. 1) which can withstand the high operating temperatures while maintaining structural integrity. Since Inconel has poor thermal conductivity the heaters are embedded in copper heat ducts that pass most of the way through the hot cap. A 0.9 mm thick disk of Inconel separates the heat duct from the heat exchanger carbon velvet while maintaining the required stiffness at the operating temperature.

The displacer is a two part design with an Inconel 625 hot flexure bounding the expansion volume and a stainless steel flexure bounding the compression volume. The hot displacer flexure is 110 mm in diameter while the cold flexure is 106 mm in diameter. The two flexures are rigidly held 49 mm apart by Inconel pins such

that they move in unison. Tuning masses are added to the displacer prior to final assembly to adjust the resonant frequency of the structure which sets the phase of the displacer relative to the diaphragm during operation. The displacer drive is additionally adjusted by tailoring the flexure profiles to achieve different effective displacer areas on the compression and expansion sides. The displacer is completely driven by gas dynamic forces as are displacers in free-piston Stirling engines. Correctly calculating the displacer operating point for this engine requires taking into account the inertia of the working gas which leads to additional pressure phase shifts between compression and expansion chambers.

Aerogel insulation fills the volume between the hot and cold displacer flexures as well as between the hot cap and pressure vessel dome to minimize parasitic heat conduction to the cold parts of the engine.

Fig. 2 shows a sectioned rendering of the engine. The dome diameter is about 30 cm and total mass is approximately 80 kg although this is much more material than strictly necessary. The alternator is three times oversized for this engine and the base plate (labeled 100 in Fig. 1) is much thicker than required. The Inconel hot cap was designed to conveniently house the cartridge heaters and is also considerably thicker than structural requirements dictate. The mass could easily be cut in half with some engineering emphasis on system mass. A nearly completed design of a new engine has a mass of about 30 kg for a design power out of 1 kW corresponding to a power density of 33 W/kg.

Unlike in the schematic drawing of Fig. 1, the hot cap in this prototype is not part of the pressure vessel and the whole engine is enclosed by a pressure vessel dome (on the left) that remains at near ambient temperature. The hot cap and copper heater ducts are visible near the center of the dome surrounded by insulation (white) in Fig. 2. The linear alternator is shown on the right. Diaphragm, spring tube and drive shaft are between the hot cap and alternator. The chambers and heat exchangers are too small to see in any detail in this scale rendering.

Temperatures in the hot side of the engine are measured using K-type thermocouples. These are located at the center of the hot cap, the center of the hot displacer flexure, next to one cartridge heaters and near one of the hot heat exchangers. In the cold side of the engine temperatures are measured with thermistors at the center and edge of the displacer flexure, and at the cold heat exchanger. Piezo-resistive pressure transducers located at the centers and edges of the hot and cold displacer flexures are used to monitor the acoustic pressure amplitude. The motions of the diaphragm, displacer, alternator reciprocator and housing are measured with accelerometers. The action of the spring tube is tracked using a strain gauge. The data acquisition system is capable of measuring and logging the amplitudes and relative phases of all these sensors simultaneously.

Since this prototype is electrically heated, the heat delivered to the engine is easily measured and logged. The heat extracted by the cooling fluid is also logged. The temperature of the cooling fluid is determined using thermistors at the inlet and outlet ports of the engine. The fluid then passes through an electrically-heated heat exchanger of known power and the outlet temperature measured with another thermistor. The heat extracted by the fluid from the engine is then determined by comparing the temperature increase through the engine in comparison to the increase due to the electrical heater.

The drive shaft is fastened to the reciprocator of a small-amplitude flux switching linear alternator. Alternator output voltage is about 160 V amplitude at full engine stroke and is characterized as an almost pure sinusoid at the engine operating frequency of around 500 Hz. The output of the alternator is connected in series with a tuning capacitor and the secondary taps of a 400 Hz variable transformer. The primary taps of the variable transformer lead to a

54 Ω power resistor. The capacitor serves to remove the reactive component of the AC signal from the alternator. The variable transformer–resistor combination behaves like a variable power resistor with a range of 20–54 Ω which in turn acts as a variable mechanical load on the engine. The engine diaphragm stroke is controlled by adjusting the electrical load. Mechanical damping of the engine varies as the inverse of the electrical resistance. It is however losses that increase faster than the square of the diaphragm amplitude (minor and join losses [9]) and the depression of temperature gradient in the regenerator with increasing stroke due to the finite thermal conductivity of the heat exchangers that determine and keep stable the actual operating point for a given electrical load resistance [5].

3. Theory/calculation

From a thermoacoustic perspective engine operation is understood as an acoustic power flow in a loop. This is illustrated schematically in Fig. 3. Out of phase motion of the diaphragm (300) and displacer (301) produces acoustic power (305) travelling towards cold heat exchanger (302). The acoustic power flowing through the regenerator (303) to the higher temperature of hot heat exchanger (304) is amplified to larger acoustic power flow (306) out of the hot heat exchanger. This larger power is absorbed by the motion of the displacer (313) and returned to the compression chamber since both flexures of the displacer are rigidly mechanically coupled together. The extra power above what is necessary to sustain the process is removed by the alternator load connected to the diaphragm and represents the mechanical output power of the engine (312). Various loss mechanisms (307, 308, 309, 310, 311) reduce the acoustic power flow around the loop leading to engine efficiency lower than Carnot efficiency for a real engine. In the compression and expansion chambers the losses are primarily relaxation loss, which is loss due to cyclic heat exchange between the gas and the walls, while in the regenerator the loss is dominated by viscous flow losses with relaxation loss very small. In the heat exchangers the viscous and relaxation losses are of similar magnitudes. In addition to these linear losses there are also

non-linear local losses at interfaces between elements. All these loss mechanisms are discussed in detail by Swift [9].

The acoustic power flows and losses in all segments of the engine loop of Fig. 3 are modeled using standard thermoacoustic methods [9,14]. In the model, one-dimensional numerical integration of acoustic propagation along the acoustic power flow direction is performed starting at the center of the compression chamber and ending at the center of the expansion chamber. The boundary conditions for the integration are that the volumetric flow must be zero at the ends. The calculation starts with zero flow and guesses an initial pressure amplitude and phase. It then performs the integration to the far end with flows added and subtracted by the motions of the flexures and reductions in flow and pressure due to various losses accounted for step by step. After the first integration the calculation proceeds to adjust the initial pressure amplitude and phase until subsequent integrations result in zero flow at the far end at which point a self-consistent solution has been found. The freely available DeltaEC [14] thermoacoustic modeling software works this way but was not used to model this engine since it does not have available axisymmetric elements with radial flow and moving walls in its library of segments. Instead, the simulation was done with custom software using the same underlying principles. Thermoacoustic simulation results of the engine are shown in Fig. 4.

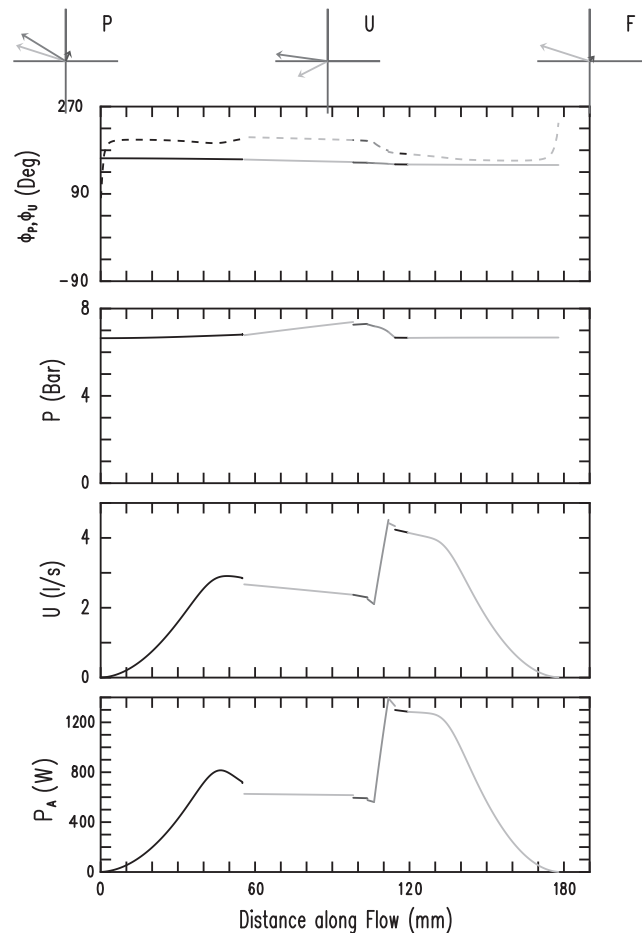


Fig. 4. Thermoacoustic simulation of the diaphragm engine. The calculated acoustic power P_A , volumetric flow amplitude U , pressure amplitude P and the phases of the pressure ϕ_p (solid) and flow ϕ_u (dashed) are shown as a function of distance along the acoustic path in the engine. Phasor diagrams at top show the pressures and flows at the ends of the acoustic path and the resulting forces on the diaphragm and displacer.

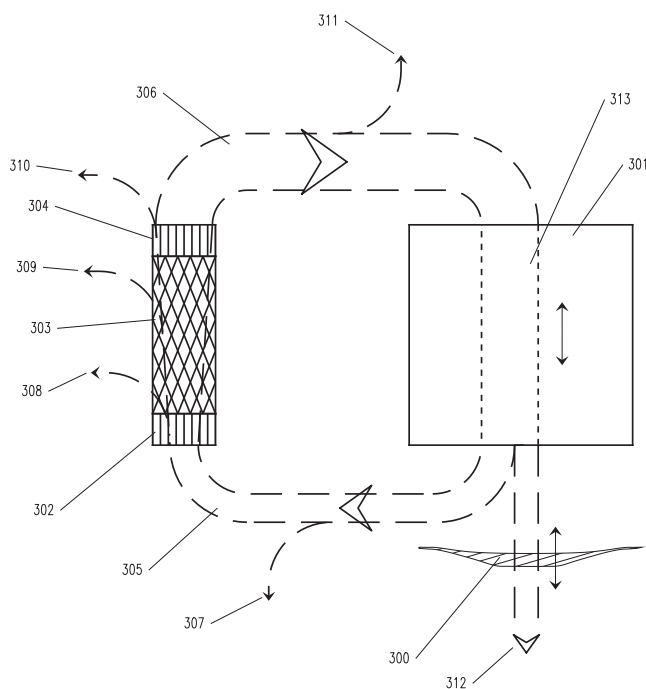


Fig. 3. Engine operation from a thermoacoustic perspective.

The acoustic power level in the acoustic segments that have non-zero length in the flow direction is shown in the bottom graph of Fig. 4. The position along the acoustic path between the compression chamber and expansion chamber is the quantity on the horizontal axis. Segment boundaries are depicted by changes in shade. The first segment (from the left between 0 and 55 mm) is the compression chamber with the flow starting at zero in the middle of the chamber. The volume flow rate increases with radius due to the motion of the diaphragm and displacer leading to growing acoustic power. The next segments have near constant acoustic power and correspond to the access tubes (55–95 mm), cold heat exchanger manifold and cold heat exchanger, in sequence with the later segments very short in spatial extent and thus hard to locate on the figure. The segment with rapid increase in volume flow rate and hence also the acoustic power corresponds to the regenerator segment which amplifies the acoustic power due to the temperature difference between compression and expansions ends. After the hot heat exchanger and hot manifold which are very short in spatial extent and thus hard to isolate on the figure, the acoustic power is absorbed in the expansion chamber (right most segment between 120 and 175 mm) by the motion of the expansion side displacer flexure.

The middle graphs of Fig. 4 show the volumetric flow and pressure amplitudes as a function of location along the acoustic path. These amplitudes and phases are calculated values from the thermoacoustic model with the input to the model the geometry of the components, the mean He working pressure, the operating frequency and the desired motion of power and displacer flexures. The top graph shows the pressure (solid) and flow (dashed) phase relative to the diaphragm motion phase. As expected, the volume flow increases through the regenerator and the acoustic oscillation phasing is predominantly travelling wave. The small discontinuities in pressure and flow between the distributed segments are due to the presence of local and join loss segments [9] in the simulation that are considered to have no spatial extent.

At the very top of Fig. 4 are shown three phasor diagrams of the pressure (left) and flow (middle) at the outer edge of the compression and expansion chambers as well as the net forces on the diaphragm and displacer (right). The phases are displayed relative to the diaphragm motion phase which is defined to lie along the positive x-axis. In the pressure and flow phasor diagrams the lighter shade is the compression chamber phasor while the darker shade is the expansion chamber phasor. The short arrow represents the pressure difference phasor which acts on the displacer as part of the displacer drive. The force phasor diagram shows the net force on the diaphragm (longer) and on the displacer (shorter). The diaphragm net force has a component along the vertical axis which is the direction of the diaphragm velocity phasor (not shown). This projection of the force thus corresponds to positive power output. The projection of this phasor along the horizontal axis in the negative direction indicates a gas spring contribution to the forces on the diaphragm which must be taken into account when calculating the diaphragm dynamics. The net force on the displacer is a delicate balance between large opposed pressure forces in the compression and expansion chambers. The engine is designed to have only a very small drive component along the displacer velocity phasor which is at an angle of about 50–60 degrees relative to the diaphragm velocity phasor. The working gas pressure difference across the displacer produces a force that acts on the displacer. A component of this force acting in the direction of the displacer velocity phasor provides the required displacer drive. There is also a perpendicular component acting in the direction of the motion phasor. If this component is negative, as is the case for the diaphragm, then it corresponds to a force which may be understood as a gas spring component. However, for the displacer the force phasor projection onto the displacer motion phasor is positive

and thus may be understood as an extra effective displacer mass, which must be taken into account to correctly calculate the displacer dynamics. Note that this effective mass is significantly larger and is not the gas flow mass which is implicitly taken into account in the thermoacoustic theory. The extra displacer effective mass is only indirectly related to the gas inertia in the sense that the inertia is partly responsible for the phase shift in the pressures acting on the two sides of the displacer.

4. Results

The engine was designed using the principles outlined above and then built and tested. The engine is essentially a mechanical simple harmonic oscillator with internal gain provided by the regenerator when a temperature differential exists across it. In the absence of a temperature gradient across the regenerator, it behaves as a slightly damped oscillator with the losses due predominantly to working gas viscosity and thermal relaxation losses. These losses can be evaluated by driving the alternator with a signal generator and sweeping the frequency to extract an oscillator quality factor Q . This was performed while varying the working gas pressure and then compared with simulation predictions. The results of these measurements are shown in Fig. 5. The Q dependence on helium gas pressure is predominantly due to displacer and diaphragm dynamics in response to changing gas pressure. The diaphragm and displacer are independent mechanical oscillators and before pressurizing the displacer resonance is near 700 Hz while the diaphragm resonance is near 400 Hz. Furthermore, the displacer unlike the diaphragm is not directly driven and thus the displacer amplitude is not locked to the diaphragm amplitude. Pressurizing the engine increases the diaphragm resonant frequency since the gas forces on the diaphragm are spring like but decreases the displacer resonant frequency since the gas forces on the displacer are mass like. The frequencies of the diaphragm and displacer overlap near 50 bar. As a result, near

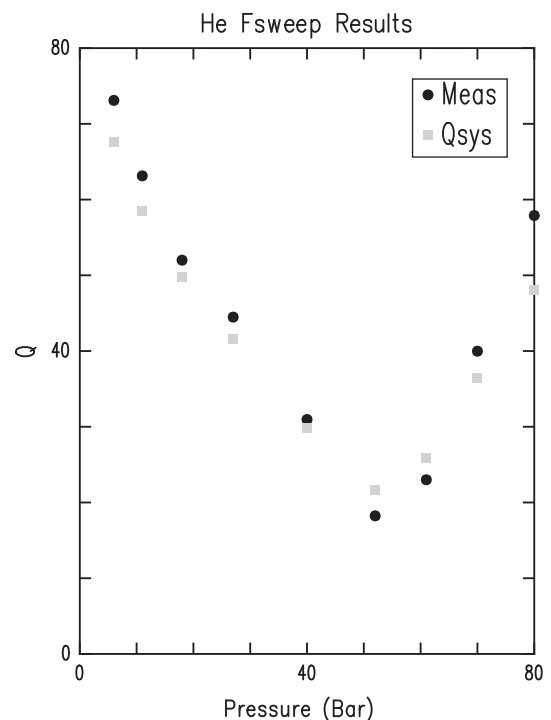


Fig. 5. Measured and calculated Q_s as a function of working gas pressure at ambient temperature.

50 bar displacer motion reaches a maximum, the gas flow through the regenerators reaches a maximum and thus viscous losses dominate at this pressure producing a minimum in the Q . These results indicate excellent agreement between prototype and simulation which, before heating, constitutes a purely acoustic simulation. It can thus be concluded that acoustic losses in the prototype are accurately modeled.

Applying heat to the hot heat exchangers, thereby producing a thermal gradient across the regenerator, produces an internal gain mechanism. At the threshold gradient, the gain matches the losses, and the engine spontaneously starts to oscillate. For this engine the threshold hot heat exchanger temperature was found to be about 80 °C above the temperature of the cold heat exchangers which were held at near ambient temperature with a circulating cooling fluid. This engine is self-starting once the threshold temperature has been reached but may be held off by providing additional external damping (for example shorting the alternator coils).

The engine is equipped with pressure sensors that measure not only the average pressure of the working gas but also the pressure swing and phase relative to the diaphragm. When driving the engine with the heat exchangers at ambient temperature, the pressure amplitude is a measure of the working volume of the engine, provided the effective area and motion of the diaphragm is known. The effective area of the diaphragm is determined by FEA calculation as well as from independent simple gas spring measurements where the gas spring volume was accurately known. From these measurement results it was apparent that the actual engine working volume was somewhat larger than initially estimated from the dimensions of the parts before assembly. There was however considerable uncertainty in the actual chamber height on the expansion side as there was substantial observed distortion due to welding the hot displacer flexure to the hot cap. After welding an internal dimension measurement was no longer possible but it was clear that the internal volume would be larger based on the external dimensions of the hot cap. Thus, in the simulation the expansion chamber height was increased from 0.6 to 1.0 mm to match the simulated compression to the measured compression. In effect the expansion chamber height was used as a single adjustable parameter in the model in order to get better agreement between the model and measurement. While it is clear that there is additional working gas volume in the engine as built, it is not possible to uniquely determine from the thermoacoustic model where the extra volume is located as essentially equally good agreement can be obtained by moving the extra volume to the compression chamber or elsewhere in the engine. Nevertheless, given the observed welding distortion, it is most likely that the extra volume is in the expansion chamber.

Fig. 6 shows a comparison of the measured pressure amplitude (symbols) to the results of the model calculation (solid lines). The lower graph displays the measured pressure amplitude and the upper graph the phases relative to the diaphragm motion phasor. Note that there are four pressure sensors, cold center (Pcc), cold edge (Pce), hot edge (Phe) and hot center (Phc). Phe was found to be non-functioning after engine assembly and data from this sensor is thus not available. The remaining three working sensors show excellent agreement with simulation after adjustment of the chamber height parameter. The data in Fig. 6 were acquired with the engine running at relatively low power with the hot exchangers at ~300 °C. The working volume conclusion is not affected by this as measured temperatures are an additional input to the simulation. The kink in the curve at around 140 μm diaphragm motion amplitude is due to an increase in mean working pressure from about 70 bar to 88 bar at this point in the experiment. The dominant effect of changing the pressure is to change the displacer resonant frequency and hence the phase angle and amplitude of the displacer motion relative to that of the

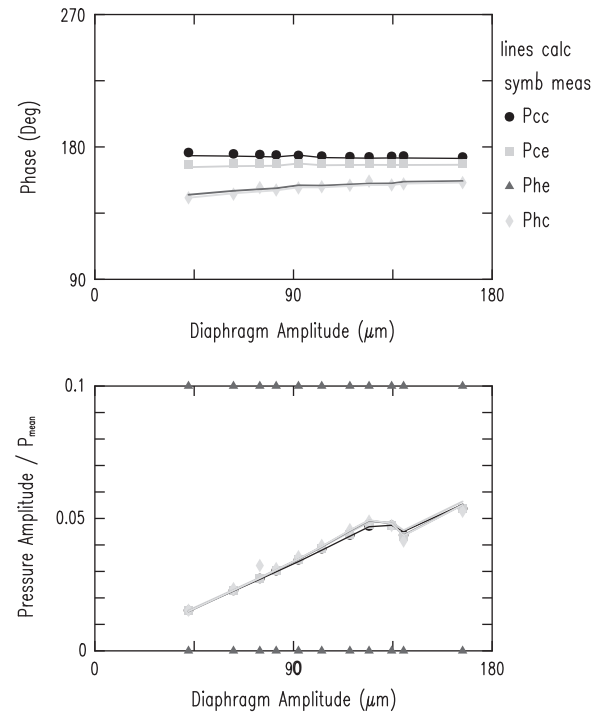


Fig. 6. Measured pressure sensor results compared to calculated pressure amplitudes and phases.

diaphragm. In this case these changes resulted in less acoustic power entering the regenerator, thus less acoustic amplification and hence smaller pressure amplitudes. The measured mean pressure was also an input to the simulation calculation which uses as inputs the measured hot and cold side temperatures, the measured diaphragm and displacer motions, the measured mean pressure, the working gas type and the geometrical model of the engine. As seen from the results, simulation and experiment are in excellent agreement.

Several experiments were conducted running the engine at a variety of diaphragm amplitudes and temperatures. Drive shaft force was determined from the product of the measured center of the diaphragm pressure oscillation amplitude and phase (Pcc) and the known diaphragm effective area. Engine net mechanical power output was then calculated from the dot product of the drive shaft force and the diaphragm velocity determined from an accelerometer. The relative sensor phase calibration was checked at threshold since there, by definition, the net power output must be zero. By threshold is meant the point at which the engine just starts running without an external load attached meaning that the diaphragm motion is large enough to reliably measure the frequency, phase and amplitude with the accelerometer mounted on the diaphragm. In this case this corresponds to an amplitude of about 1 μm in contrast to the full stroke amplitude of 200 μm . Power is proportional to the square of the amplitude and this level of motion corresponds to an insignificant power level. However, the gains must match the losses in order to have self-sustained oscillation and the net power out is zero as there is no external load. Thus, at threshold the force phasor must be 180 degrees out of phase with the diaphragm motion phasor and thus 90 degrees out of phase with the diaphragm velocity phasor so that the dot product of force and velocity is zero. Based on this condition, it was found that the pressure phase needed to be adjusted from the initial calibration by around 1 degree. This process in effect subtracts the small pure mechanical losses of the mechanical oscillator system formed by the diaphragm, spring tube, and attached alternator moving mass. Without this correction the

measured power is indicated power but after this correction, subsequently applied to all measurements, the measured power is the net mechanical output power. This calibration step also removes any small systematic phase calibration errors from the sensor signals that may be present. Lastly, this calibration accounts for any small pressure phase changes center to edge across the diaphragm. The correction works provided the compression space center pressure sensor phase angle relative to the horizontal axis remains relatively close to 180 degrees and results in less than 1% error in calculated net power for Pcc phasor angles with up to 25 degrees deviation from 180 which was satisfied in all experiments.

The performance of the engine as a function of hot exchanger temperature is shown in Fig. 7. Results are from a set of experiments done on different days and with different alternator builds. Some scaling was applied to all but the highest temperature point to normalize for diaphragm and displacer stroke differences but the last point at the highest temperature is as measured. This last point was measured with the hot heat exchangers near 500 °C and the cold heat exchangers near 30 °C and the diaphragm motion amplitude at the maximum design amplitude of 200 μm . Plotted along the horizontal axis is the diaphragm motion amplitude relative to the engine housing, determined from the vector difference of the diaphragm and housing accelerometer phasors. The displacer motion relative to the housing is determined in a similar fashion and both results used as inputs to the model. The simulation results in the plot were calculated using the measured motions for the highest temperature point as input and varying only the hot exchanger temperature to produce the gray points. The plotted measured output powers are the net mechanical powers determined using the procedure discussed above. The efficiencies are net mechanical efficiencies determined by dividing the net mechanical powers by the measured input electrical heater powers. The highest temperature point corresponds to a measured net power of 585 W at a net efficiency of 21%.

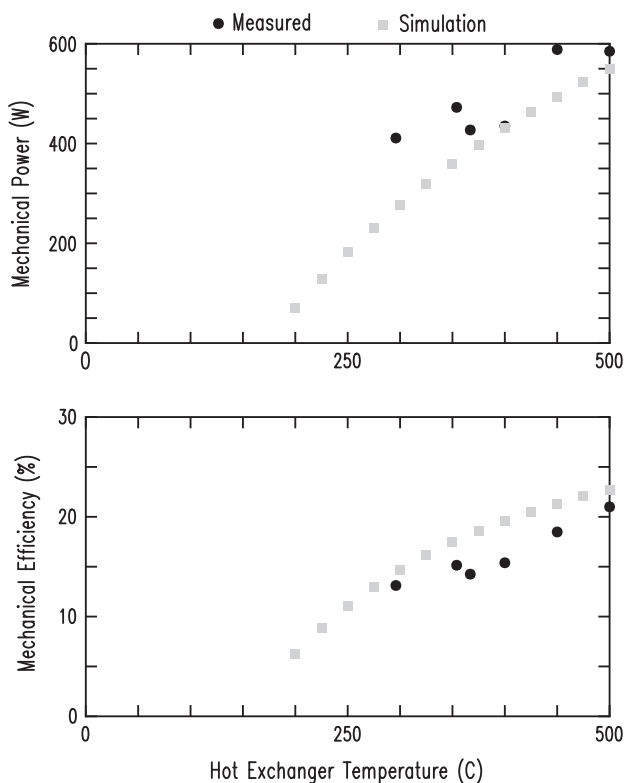


Fig. 7. Engine performance results vs. hot exchanger temperature.

5. Discussion

As can be seen in Fig. 7 the measured power is consistently larger than the power expected from the simulation but the efficiency is less. At this point the reason for the discrepancy in efficiency and power is not known. Given that the ambient temperature Q results and gas pressure oscillation results match the simulation very well, the acoustic losses are most probably modeled correctly. It is therefore likely that the lower efficiency is due to some unintended streaming flow between the separate regenerator modules or extra regenerator inefficiency loss that causes extra heat to be transferred from the hot to the cold heat exchangers.

In this prototype, the displacer amplitude was only about 80% of the design goal and the displacer phase angle was about 10 degrees larger than intended. It should therefore be possible to extract more power and efficiency from an engine of the same form factor. Displacer drive is predominantly due to a delicate balance of large, nearly equal and opposite, gas pressure forces acting on the compression and expansion sides of the displacer. Small errors in calculating these pressures have a relatively large effect on displacer drive and phase. Fortunately, the engine proved to be remarkably forgiving, working over a very wide parameter space. However, achieving optimum performance will likely require some experimental fine tuning of displacer flexure profiles and mass in order to get closer to the intended displacer dynamics. This does not affect the comparison between measurement and simulation in Fig. 7 as the actual measured displacer motion was used as an input to the simulation.

The linear alternator performance proved to be disappointing. By comparing the measured net mechanical power to the measured alternator electrical output power it was determined that the alternator was about 85% efficient at small stroke but dropped to about 60% efficiency at full stroke. It thus has significant non-linear losses that grow rapidly with increasing stroke. At least part of the problem are losses due to squeezing air from between the pole gaps. A new alternator design, currently under construction, will address this problem by running the alternator in vacuum.

It still remains to be demonstrated that heat can be supplied from an external source such as a burner rather than from internal electric cartridge heaters. This is a non-trivial task but progress is being made on this front. Reports on progress with external heat input may be expected in the near future.

6. Conclusions

To our knowledge, this engine has the highest efficiency and power output of any pure diaphragm engine constructed to date. The engine functioned as intended with dynamics close to model predictions. Thus, successful operation at high pressure and high frequency has been proven. Agreement between thermoacoustic simulation results and measured performance was adequate to give confidence that future engines designed using this method will perform as intended. In the design process this engine was optimized more for power than for efficiency. Future engines designed more for efficiency can thus expect to achieve greater efficiency than was demonstrated with this engine. Additional engines incorporating some changes in design are currently under construction and are expected to produce more power and higher efficiency. The full parameter space of pressure, frequency, diaphragm size and stroke has not been explored in detail theoretically particularly given the effects of these parameters on the non-thermoacoustic aspects of the engine such as pressure vessel mass, alternator efficiency and heat exchanger temperature deltas. An optimal engine for a specified prime performance metric may well lie elsewhere in the available parameter space. This engine

is merely an example of choices in the parameter space that look promising and may lead to a potentially commercially successful Stirling/thermoacoustic Engine.

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